SKOGLUND

Design of a 6 Horse Power 2 Cylinder Gasolene Engine Air Cooled

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DESIGN OF A 6 HORSE POWER 2 CYLINDER GASOLENE ENGINE AIR COOLED

BY

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Carl August Skoglund

ENTITLED Design of a 6 Horse Power 2 Cylinder Gasolene Engine Air Cooled

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE OF Bachelor of Science in Mechanical Engineering

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PLATES AND TABLES

DESIGN OF A 6 HORSEPOWER 2 CYLINDER GASOLINE ENGINE.-- AIR COOLED.

The above subject for a thesis was propossed for several reasons. It was desired by the Director of the Shop Laboratories to have two engines, one water-cooled and one air-cooled, so designed that all parts would be the same except the cylinders. With two such engines it would be possible to test the relative merits and efficiences of water-cooled and air-cooled types. It was desired also th have a design of a practical, up to date engine which could be made in the shops by the University students. The design of the water-cooled type was allotted to one student and the air-cooled to another, but the other details were to be designed in common.

ARTICLE I.- SPECIFICATIONS - The following specifications for the design were given:-

Type of engine - Four stroke cycle.

Number of cylinders - Two - Vertical.

Horsepower - Six.

Nominal Speed - 600 Revolutions per minute.

Valves - Enclosed.

Oiling - Splash system.

Water circulating pump - centrifugal type.

Cylinders cast en bloc.

Commecting rod - cast manganese bronze.

ARTICLE II.- PISTON DISPLACEMENT - LEVIN'S METHOD.- The first step in the design is to determine the piston displacement necessary to give the required power. No matter what method is used to calculate the piston displacement, several assumptions must be made.

Using the method given in the "Modern Gas Engine and the Gas Producer" by Levin, the following notation will be used:-

V_a = specific volume of final charge after the completed suction stroke.

V_o = specific volume at standard pressure and temperature.

T_o = standard temperature 460 62 522 abs.

T_a = 460° 86°F 546 abs. See Levin p. 126-128.

H = heating value in B.t.u. per cu. ft. of gasoline vapor.

D = suction displacement in cubic feet per minute per brake horse power.

E = theoretical efficiency $= 1 - \frac{1}{r^{n-1}}$.

r = compression ratio.

n = ratio at specific heats c_p/c_v assumed 1.28

Efy = practical thermal efficiency based on indicated horse power.

$V_a/V_o = P_o/P_a \times T_a/T_o = 14.7/13.7 \times 546/522 = 1.12$

The rather high value of 13.7 lb. per sq. in. for the initial pressure is taken because exceptionally large values are to be used in the design. Also the piston speed will be low. a vol. of air required by analysis per cubic foot of gasoline vapor 47.14 cu. ft. 15% excess 7.07 Cu. ft. ax total vol. of air per cu. ft. 54.21 cu. ft.

Then the volume of the expanded normal charge containing one cubic foot of gasoline vapor is $V_a/V_o(ax+1) = 1.12 \times 55.21 =$

61.84 cu. ft. The specific volume of gasoline vapor is given as 4.2 cu. ft. per lb.; and using 18,500 B.t.u. per pound as the heating value of gasoline, the heating value per cubic foot of gasoline vapor becomes, 18,500/4.2 4,400 B. t.u. The heating value per cubic foot of expanded normal charge will be,

$$\frac{H}{V_a/V_0(ax+1)} = \frac{4,400}{61.84} = 71.2 \text{ B.t.u.}$$

Assuming a compression ratio (i.e. the ratio of total volume of cylinder to the clearance volume) of 4.5 and mechanical efficiency of 0.80, the suction displacement per minute per brake horse power will be

$$D = \frac{42.42}{0.80 \times \text{Efy.} \dfrac{H}{V_a/V_0(ax+1)}} = \frac{42.42}{0.80 \times 0.255 \times 71.2} = 2.92 \text{ cu. ft. per}$$

min. Then $\frac{3 \times 2.92 \times 1728}{300} = 50.5$ cubic inches piston displacement for one cylinder.

A bore and stroke of 3 5/8" × 5" gives 51.7 cu. in. The m.e.p. will be 5.4 × 0.255 × 71.2 = 98 lb. per sq. in.

When the cylinder and compression space were laid out on the drawing board, the compression ration was found to be 4.47

Efy corresponding to this ratio is 0.216.

D = 0.255/0.216 × 2.92 = 3.45 cu. ft. per min.

The piston displacement is 0.255/0.216 × 50.5 = 59.6 cu. in.

The m.e.p. = 5.4 × 0.216 × 71.2 = 83 lb. per sq. in.

A bore and stroke of 3 3/4" × 5 3/8" would give 59.41 cu. in. piston displacement.

GULDNER'S METHOD. Guldner in his book derives the following

formula for the diameter of cylinder for gasoline engines:-

$$D=\sqrt{\frac{108NL}{\underline{n}HSn_w n_e}}$$ in feet, in which

N = nominal brake horse power per cylinder.

$\underline{n}$ = R. P. M. 600

H = heating value per lb. gasoline = 18,500 B.t.u.

L = actual amount of air required per lb. gasoline.

S = stroke in feet---assumed when solving for D.

n_w = thermal efficiency = 0.16

n_e = volumetric efficiency = 0.80

The nominal brake horse power N includes a 20% overload capacity. The design under consideration is to be rated at its maximum brake horse power. If N is made equal to three then the maximum B.h.p. would be 1.2 × 3 = 3.6 H.p. Therefore the N to use in the formula is 3/3.6 × 3 = 2.5. The value of L may be taken as 15% in excess of the theoretical amount of air or 1.15 186. Whence L 214 cubic feet per pound of gasoline. Assuming S = 5" or 0.416 feet, n_e = 0.80, and n_w = 0.16

$$D=\sqrt{\frac{108\times2.5\times214}{600\times18,500\times0.416\times0.16\times0.80}}=\sqrt{0.0979}=0.313 \text{ feet.}$$

D = 3.755 inches.

S = 5 inches.

3 3/4" × 5" gives 55.225 cu. in. piston displacement. The stroke-bore ratio for this cylinder is 5/3.75 = 1.33.

The 3 3/4" × 5" cylinder will be used although not as conservative as that of 3 3/4" × 5 3/8" which was obtained by using Levin's method, because it is desired to have a small, light engine. If necessary the speed may be increased somewhat above 600 R.P.M.

ARTICLE III.- VOLUME OF COMPRESSION CHAMBER.- The volume of the compression space required to give a compression ratio of 4.5 may be obtained as in the following:-

Let V_1 = total cylinder volume and V_6 = volume of the compression space. Then the compression ratio is $V_1/V_6 = r$.

$V_1/V_6 = 4.5$

$V_1 = 4.5V_6$

$V_1 - V_6$ = piston displacement.

$V_1 - V_6 = 4.5V_6 - V_6 = 3.5V_6$

The ratio of the compression space or volume to the piston displacement is $V/V_1 - V_6$ or $1/3.5 = 0.286$. The required volume will then be $0.286 \times 55.225 = 15.79$ cubic inches.

The outline of the compression space or compression chamber will be as shown in Fig. a. The valves will be in the head.

The volume of the cylinder is $\pi r^2 h = 17.28$ cubic inches. The volume occupied by the piston when projecting 1/8 inch above the bore is 1.38 cubic inches. The net compression volume is then 17.28 - 1.38 or 15.9 cubic inches. Then the actual compression volume in percent of piston displacement is $\frac{15.9 \times 100}{55.225} = 28.806\%$.

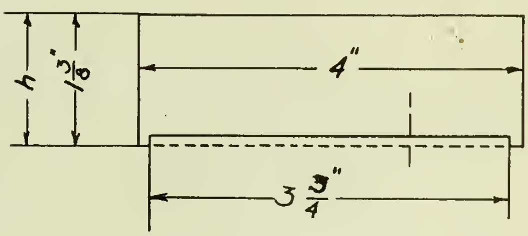


FIG. a

ARTICLE IV.- PRESSURE-VOLUME RELATIONS.- The pressure on the piston at several points along the stroke must be calculated

for the compression and expansion curves so that a probable indicator card may be plotted with the values obtained.

The value of n_c, or n for the compression curve will be assumed 1.28, and n_e for the expansion curve will be taken as 1.33. Some writers assume for these, the values 1.3 and 1.35 respectively.

The equation for the compression curve is $p_1 V_1^{1.28}$ = constant, and for the expansion curve $p'_1 V_1^{1.33}$ = constant. Knowing the bore and stroke of the engine, the volumes for different positions of the piston may be easily figured.

V_1 = total cylinder volume = piston displacement + clearance

V_6 = compression volume		= 15.905
$V_2 = V_1 - 1/5 \times$ piston displacement	= 71.13 − 11.045	= 60.085
$V_3 = V_1 - 2/5 \times 55.225$	= 71.13 − 22.09	= 49.040
$V_4 = V_1 - 3/5 \times 55.225$	= 71.13 − 33.135	= 37.995
$V_5 = V_1 - 4/5 \times 55.225$	= 71.13 − 44.180	= 26.950

These volumes then are the volumes included between the piston head and the cylinder head when the piston is at the relative positions along its stroke as indicated in Fig. b.

ARTICLE V.- COMPRESSION CURVE.- The pressures along the compression curve at these points are,

p_1 = 13.7 lb. per sq. in. absolute,

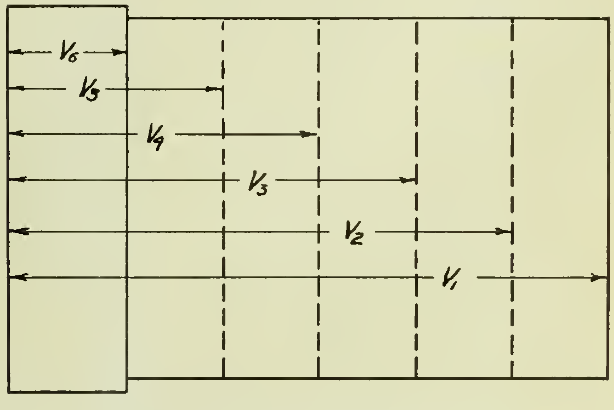


FIG. b

$$p_2 = p_1(v_1/v_2)^n = 13.7(71.13/60.085)^{1.28}$$
$$p_3 = p_1(v_1/v_3)^n = 13.7(71.13/44.04)^{1.28}$$
$$p_4 = p_1(v_1/v_4)^n = 13.7(71.13/37.995)^{1.28}$$
$$p_5 = p_1(v_1/v_5)^n = 13.7(71.13/26.95)^{1.28}$$
$$p_6 = p_1(v_1/v_6)^n = 13.7(71.13/15.905)^{1.28}$$

$p_1 = 13.7$ lb. per sq. in. abs.

$\log 1.183 = 0.073$; $\times 1.28$; $0.0935 \equiv 1.242$	Therefore $p_2 = 13.7 \times 1.242 = 17.3$ lb. per sqq in. abs.
$\log 1.45 = 0.161$; $\times 1.28$; $0.206 \equiv 1.61$	Therefore $p_3 = 13.7 \times 1.61 = 22.1$ lb. per sq. in. abs.
$\log 1.872 = 0.271$; $\times 1.28$; $0.347 \equiv 2.225$	Therefore $p_4 = 13.7 \times 2.225 = 30.5$ lb. per sq. in. ags.
$\log 2.64 = 0.421$; $\times 1.28$; $0.54 \equiv 3.47$	Therefore $p_5 = 13.7 \times 3.47 = 47.5$ lb. per sq. in. abs.
$\log 4.47 = 0.65$; $\times 1.28$; $0.832 \equiv 6.8$	Therefore $p_6 = 13.7 \times 6.8 = 93.2$ lb. per sq. in. abs.

ARTICLE VI.- PRESSURE-TEMPERATURE RELATIONS.- In order to calculate pressures for points on the expansion curve the explosion pressure must first be determined. This pressure can only be approximated, because there are several conditions affecting it which are not definitely known. It is true too that the compression pressure can not be determined accurately because the inlet pressure and the value of n must be assumed, but it can be more closely approximated than the explosion pressure. Actual tests have shown that the maximum pressures obtained are consid-

erably less than the calculated theoretical pressures. Several explanations for this discrepancy have been offered but no theory as yet accounts for it entirely. In the end it amounts to this, that not all of the heat supplied in the charge of fuel is used to raise the temperature and pressure. The amount utilized is approximately one half of the heat supplied. This value will be used in calculating the probable explosion termperature T_6'.

The initial temperature of the charge at the beginning of the compression stroke must be asumed. This will be in the neighborhood of 100° F. The specific heat of constant volume for gasoline mixture will vary with the different grades of gasoline and with the different mixtures. P. M. Heldt, in his book "The Gasoline Automobile", gives 0.20 as the specific heat. This value will be used.

Starting with an initial temperature of 100°F or 560° absolute, the temperature at compression will be,

$$T_6 = T_1(r)^{n-1} \text{ or } T_6 = 560(4.47)^{0.28}$$

$$\begin{array}{r} \log 4.47 = 0.65 \\ 0.28 \\ \hline 0.182 \end{array} \equiv 1.522$$

Therefore $T_6 = 1.522 \times 560 = 854°$ absolute.

The explosion temperature or temperature at the end of combustion will be the sum of the temperature at compression plus the increase in temperature due to the heat of combustion of the charge. The amount of heat per lb. of gasoline mixture is, 18,500/19 = 974 B.t.u., using 18 parts of air by weight to oneof gasoline. Letting T'_6 represent the temperature after combustion.

$$T'_6 = T_6 + \frac{1/2 \times 974}{0.20} = 854° + 2436° = 3290° \text{ absolute.}$$

ARTICLE VII.- THE EXPANSION CURVE.- The maximum pressure p'_6 may be obtained from the following relation of pressures and temperatures.

$$p'_6/p_6 = T'_6/T_6 \quad \text{or} \quad p'_6 = p_6 \times T'_6/T_6$$

$$p'_6 = 93.2 \times 3290/854 = 359 \text{ lb. per sq. in. abs.}$$

$$p'_5 = (V_6/V_5)^n . p'_6 = (0.590)^{1.33} \times 359$$

$$p'_4 = (V_6/V_4)^n . p'_6 = (0.419)^{1.33} \times 359$$

$$p'_3 = (V_6/V_3)^n . p'_6 = (0.324)^{1.33} \times 359$$

$$p'_2 = (V_6/V_2)^n . p'_6 = (0.2615)^{1.33} \times 359$$

$$p'_1 = (V_6/V_1)^n . p'_6 = (0.224)^{1.33} \times 359$$

p'_6 = 359 lb. per sq. in.abs

log 0.590 = 9.7705-10
1.33
1.695 ≡ 0.496, Therefore p'_5 = 359 × 0.496 = 178 lb. per sq. in. abs.

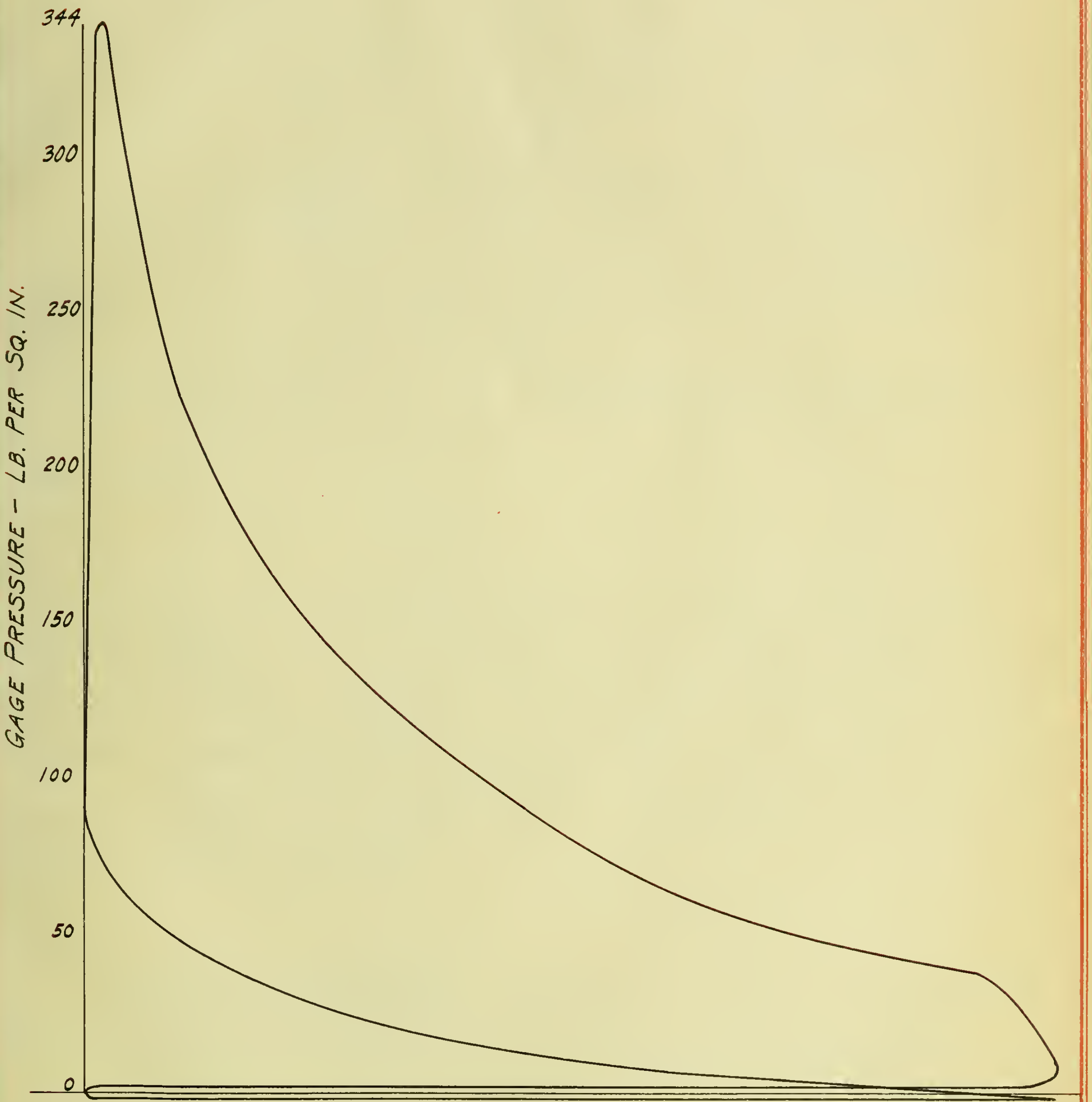
log 0.419 = 9.622 -10
1.33
1.348 ≡ 0.3145, Therefore p'_4 = 359 × 0.3145 = 118 lb. per sq. in. abs.

log 0.324 = 9.51-10
1.33
1.348 ≡ 0.223, Therefore p'_3 = 359 × 0.223 = 80.2 lb. per sq. in. abs.

log 0.2615 = 9.417-10
1.33
1.225 ≡ 0.168, Therefore p'_2 = 259 × 0.168 = 60.3 lb. per sq. in. abs.

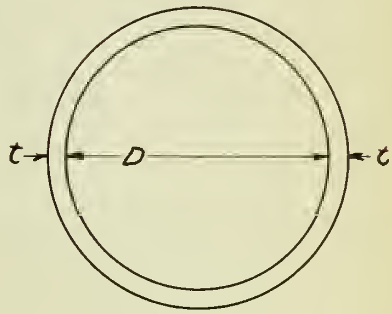
log 0.224 = 9.35 -10
1.33
1.1355 ≡ 0.1367, Therefore p'_1 = 359 × 0.1367 = 49.1 lb. per sq. in. abs.

PLATE I.- Page 10, shows the indicator card.



INDICATOR CARD
M.E.P. = 92 LB. PER SQ. IN.
COMPRESSION PRESS. = 78 LB. PER SQ. IN.

ARTICLE VIII.- THE CYLINDER WALL.- For one inch along the axis of the cylinder, the metal resisting the internal pressure is $2t \times 1$. The total force tending to part the cylinder is $P_m \times D$ lb. If S is the tensile stress in lbs. per sq. in. then $P_m \times D = 2t \times S$, P_m = max. explosion pressure. The maximum thickness that could be cast without much difficulty would be 1/4 inch, then the tensile stress would be:-



$S = \dfrac{344 \times 3.75}{2 \times 0.25}$ = 2580 lbs. per sq. in., which is a safe value for cast iron.

ARTICLE IX.- THE COOLING FLANGES.- In order to have the outside diameters of the water-cooled and air-cooled cylinders the same, the cooling flanges on the air-cooled cylinder will be limited to a radial depth of 9/16 of an inch, or the outside diameter of the flanges will be 5 3/8 inches. The spacing will be 5/16 of an inch center to center.

ARTICLE X.- THE VALVES.- Locating the valves in the cylinder head allows room for valves 1 3/4 inches in diameter, with a clear diameter of 1 1/2 ". The lift will be made 3/8" and the angle of valve seat 45°.

ARTICLE XI.- VALVE TIMING.- The timing of the valves is as

follows:-

Inlet valve opens 6^{0} after top dead center.

Inlet valve closes 16^{0} after bottom dead center.

Total opening of inlet $= 190^{0}$ degrees of crank motion.

Exhaust valve opens 34^{0} before bottom dead center.

Exhaust valve closes 6^{0} after top dead center.

Total opening of exhaust = 220 degrees of crank motion.

Figure 7 is the timing diagram.

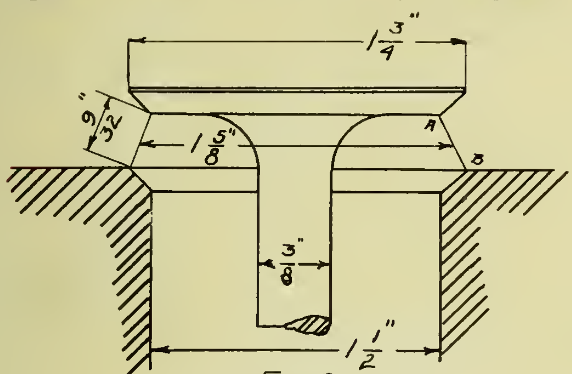


FIG. 8

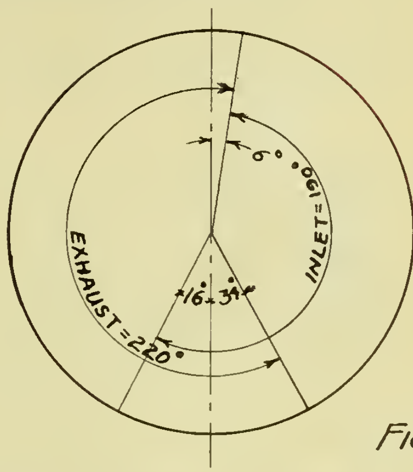


FIG. 7

ARTICLE XII.- VELOCITY OF EXHAUST GAS.- Figure 8 shows the exhaust valve and seat with the valve full open. The area under the valve is the area of the conical surface having AB as a length and 1 5/8 inches as a mean diameter. Then the total area under the valve is $1.625\pi \times 9/32$ = 1.435 square inches.

Area of 3/8" stem = 0.111 square inches.

Net area of opening = 1.324 square inches.

At 600 R.P.M. the average piston speed is 500 feet per minute. The approximate average velocity of the exhaust gas

$$V_g = \frac{500 \times 11.045}{60 \times 1.324} = 69.5$$ feet per second or 4170 feet per minute.

This low velocity is the result of using a large exhaust valve. It reduces considerably the back pressure due to the exhaust. The exhaust passage from each cylinder will be made 1 1/2 inches in

diameter. This size gives an area somewhat greater than the area of valve opening.

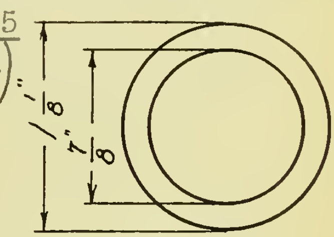
ARTICLE XIII.- THE PISTON PIN.- The piston pin may be designed for the allowable maximum bearing pressure and checked by calculating the bending and shearing stresses. An average value for the maximum bearing pressure on the piston pin in small light engines would be 2400 pounds per square inch. Lower pressures are used for heavier engines.

In this design it is desirable to keep the bearing pressure low because splash oiling will be used. The maximum bearing pressure will then be taken as 1800 pounds per square inch of projected area of the pin. The maximum load on the pin will be, $D^2\pi/4 \times 344$, and the length of pin bearing will be taken as D/2. Then the diameter d_o will be $\frac{\pi D/4 \times 344}{1800 \times D/2} = \frac{11.045 \times 344}{1800 \times 1.875}$, $d_o = 1.127$. inches or 1 1/8 inches.

In calculating the bending stress, the pin is considered as a beam uniformly loaded. The bending moment is $P^{1}/8$. The resisting moment for a hollow pin when subjected to bending is,

$\pi \frac{S}{32}\left(\frac{d_o^4 - d_i^4}{d_o}\right)$; in which d_o is the outside diameter of the pin, d_i the inside diameter, and S the stress in pounds per square inch.

Then the stress is: $S = \frac{32Pl}{8\pi\left(\frac{d_o^4 - d_i^4}{d_o}\right)} = \frac{4 \times 3800 \times 1.875}{\pi\left(\frac{1.6 - 0.585}{1.125}\right)}$.



(d_o = 1 1/8 inch)
(d_i = 7/8 inch)

S = 8940 pounds per square inch, which is considerable lower

than is used in most designs.

The shearing stress, S_s will be

$$S = \frac{3800}{2\left(\frac{\pi d_o^2}{4} - \frac{\pi d_i^2}{4}\right)} = \frac{3800}{2(0.994 - 0.60132)}$$

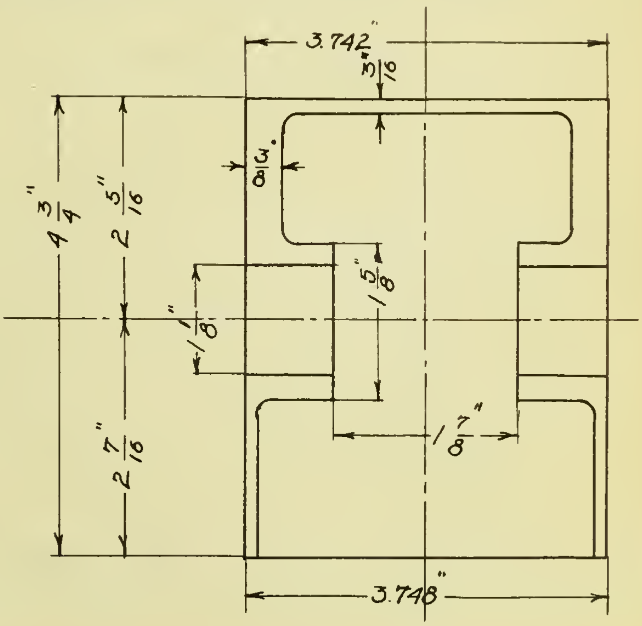
S_s = 4840 pounds per square inch.

The hollow pin of 1 1/8 inch d_o, and 7/8 inch d_i is lighter than a solid pin 3/4 inches in diameter. The cross sectional area of the pin is 0.3927 square inches. The weight is $3.5 \times 0.3927 \times 0.28 = 0.384$ pounds.

ARTICLE XIV.- THE PISTON.- The design of the piston is purely empirical. To allow for heat expansion the practice is to use 2 thousandths of an inch per inch of cylinder diameter at the head of the piston and 1/2 thousandth at the bottom end. Then the diameter at the top will be $d_h = 0.998 \times 3.75 = 3.742$ inches.

Diameter at open end $d_o = 0.9995 \times 3.75 = 3.748$ inches.

Assuming the length of piston 1.25 D, L = 4 3/4 inches. From the dimensions of the piston, Fig. c, the calculated weight was found to be 3.823 pounds.



ARTICLE XV.- THE PISTON RINGS.- The width of ring is usually made D/20. With this, w becomes 3.75/20 = 0.1875 or

3/16 inch. Heldt, in his "Gasoline Automobile" Vol. I. derives D/27.5 for the maximum thickness of the ring opposite the cut and takes D/27.5 as the thickness at the cut. Then the outside diameter $D_o = \frac{D/27.5 + D/55}{2} = 1.027\ D$. Making an allowance of 0.008D for grinding the ring blank should be turned 1.035Doutside and D - 0.027D or 0.973 D inside diameter. These dimensions are then, $1.035 \times 3.75 = 3.881$ inches outside diameter of the blank, $0.973 \times 3.75 = 3.649$ inches inside diameter of blank, and $1.027 \times 3.75 = 3.851$ inches, the outside diameter of the finished ring. The amount to be cut out of the ring on the circumference will be $3.1416 \times (3.851 - 3.75)$ plus the clearance, or 0.3173 inches+clearance. Making the cut at 45°, the width will be $0.707 \times 0.3173 +$ clearance $=$ 0.2243 +clearance. The cut will be made 1/4 inch wide at an angle of 45° which leaves a clearance of $0.0257/0.707 = 0.364$ inches along the circumference to allow for heat expansion. The thickness opposite the cut is 3.75/55 or 0.068 inches.

ARTICLE XVI.- THE CONNECTING ROD.- The connecting rod was laid out on the drawing board and the section proportioned empirically. The mid section is shown in Fig. I. The rod is considered as a column with one end fixed and one round as it is between the cases of both ends fixed or both ends round. Rankine's formula for columns is $S = P/a(1 + \phi(l/r)^2)$ in which S is the stress in pounds per square inch, P the total pressure or load on the rod at its middle, ϕ a constant depending on the material used and the condition of the

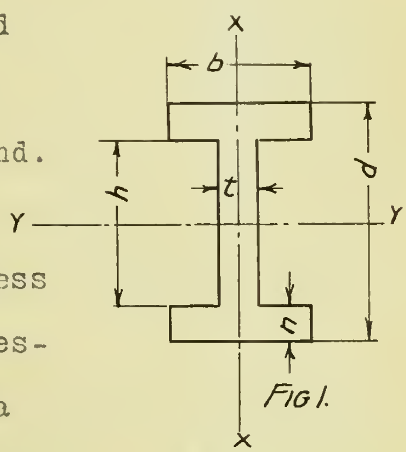


FIG I.

ends, and (l/r) the ratio of the length of rod to the least radius of gyration.

The dimensions of the section are as follows:-

area $= 0.358$ square inches.

b $= 0.688$ inches

d $= 1.188$ inches

t $= 0.188$ inches

n $= 0.156$ inches

h $= 0.875$ inches

The moment of inertia about the X - X axis $I_x = 2nb^3 + ht^3/12$, and about the Y - Y axis $I_y = bd^3 - h^3(b-t)/12$. The radius of gyration $r = \sqrt{I/a}$. Substituting the dimensions given, I_x becomes 0.00893 and $r_x = 0.158$. Also $(l/r)^2$ becomes 4850. The constant for wrought iron with one end fixed and one end round is given by Merriman as 0.0000542. The total pressure at the time of explosian is $11.045 \times 344 = 3800$ pounds. Then the stress

$S_x = \frac{3800}{0.358}(1 + 0.0000542 \times 4850) = 13,400$ pounds per square inch.

Substituting in the expression for I_y the given value, $I_y = 0.093$ and r_y becomes $= \sqrt{0.093/0.358} = 0.51$. Then $(l/r)^2$ 465 and $S_y = 3800/0.358(1 + 0.0000542 \times 465) = 11,600$ pounds per sq. in

When the whipping effect of the connecting rod is the greatest the load on the rod due to the pressure on the piston is much decreased and calculations show that the combined stress due to the whipping and the load on the rod, is less than the stress calculated for the pressure alone when it is a maximum at dead center.

The weight of the rod was estimated or calculated from the dimensions to be 4.058 pounds. The distance, d from the center of

the crank pin to the center of gravity of the rod was 3.045 inches.

ARTICLE XVII.- THE INERTIA FORCES.- The horizontal inertia forces are made up of two parts, the force required to accelerate the piston and the force required to accelerate the connecting rod. The acceleration of the piston pin or the small end of the connecting rod is $v^2/r(\cos 0 \;\; r/l\cos 20)$, in which v is the velocity of the crank pin in feet per second, r is the length of crank, and 0 the crank angle. The derivation of this expression can be found in many tests so no time will be given for that here. The radial acceleration of the crank pin is v^2/r and the hori-

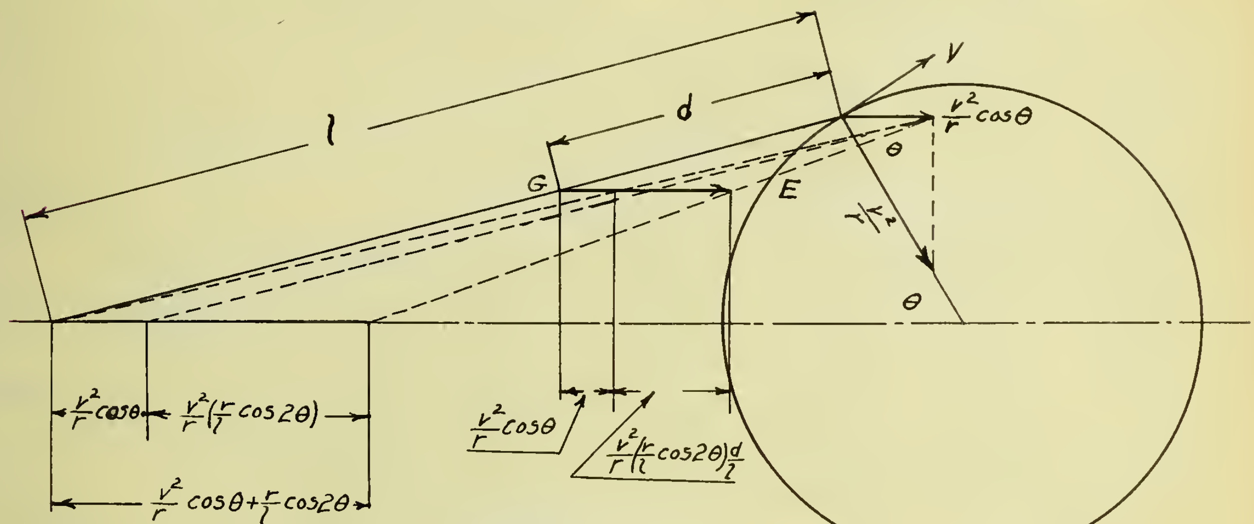


FIG. 2

zontal component is $v^2/r \cos 0$. The accelerations are shown as vectors in Fig. 2. By inspection of the figure it can be seen that the acceleration of the center of gravity G is represented by the vector GE and GE $v^2/r(\cos 0 \;\; dr/l^2\cos 20)$. The horizontal force necessary to give the connecting rod this acceleration is then $W_c/g \;\; v^2/r(\cos 0 \; dr/l^2\cos 20)$. W_c being the weight of the connecting rod and g the acceleration due to gravity. This force and

the inertia forces are identical.

The horizontal force necessary to accelerate the piston is $W_p/g \times v^2/r(\cos\theta + r/l\cos 2\theta)$. This is equal to the horizontal inertia force for the piston. The total inertia force is the sum of the two. Letting F represent the inertia force per square inch of the piston due to the piston, and F' the inertia force per square inch due to the connecting rod, the F becomes

$$\frac{0.381 \times 171.5}{32.2 \times 0.2083}(\cos\theta + r/l\cos 2\theta), \text{ and } F' = \frac{0.367 \times 171.5}{32.2 \times 0.2083}(\cos\theta + dr/l\cos 2\theta).$$

The inertia forces per square inch of piston for the different crank angles are shown in Table I. These were calculated for a speed of 600 R.P.M., which is normal speed.

The forces for each stroke of the cycle are shown plotted in a curve, Fig. 3 on Plate II, Page 19.

TABLE I.

θ	F	F'	F + F'
0	11.90	9.96	21.86
30	9.52	8.41	17.93
60	3.75	4.39	8.14
90	-2.21	-0.59	- 2.80
120	-5.96	-4.98	-10.94
150	-7.31	-7.81	-15.12
180	-5.51	-9.07	-16.58
150	-7.31	-7.81	-15.12
120	-5.96	-4.98	-10.94
90	-2.21	-0.59	- 2.80
60	3.75	4.39	8.14
30	9.52	8.41	17.93
0	11.90	9.96	21.86

Figure 3 on Plate II, Page 19, shows the gas pressure and inertia forces for one cylinder thru a complete cycle of four strokes or two revolutions of the crank. The net force in pounds per square inch acting on the piston is obtained from the resultant curves in each stroke. Multiplying the net force on the piston for each 15° of the crank travel by the piston area gives the total force P_R. These are shown in Table II, Page 21.

ARTICLE XVIII.- TURNING MOMENT.- Figure 4 on Plate II, Page 20

shows the turning moment curves, which were plotted from the values shown in Table II. The turning moment for any crank angle 0 is the product of the tangential force by the crank length in feet. The tangential force is the tangential component of the total thrust Q transmitted thru the connecting rod to the crank pin. The thrust Q is the resultant of P_R and P_g, the pressure on the piston and the guide pressure. It will be accurate enough to assume $Q = P_R$. The tangential factor multiplyed by the pressure P_R gives the tangential force.

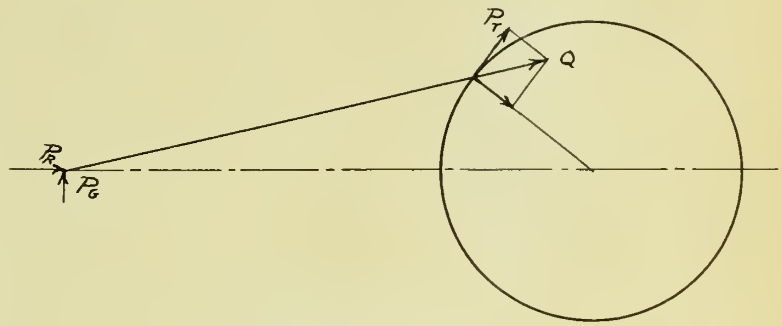


FIG. 5

The following table shows the total net pressure on the piston, P_R; the tangential factor; the tangential force; and the turning moment.

The expression for the tangential factor is,

$$\left(\sin\theta + r/l\frac{\sin\theta\cos\theta}{1-(r/l)^2\sin^2\theta}\right).$$

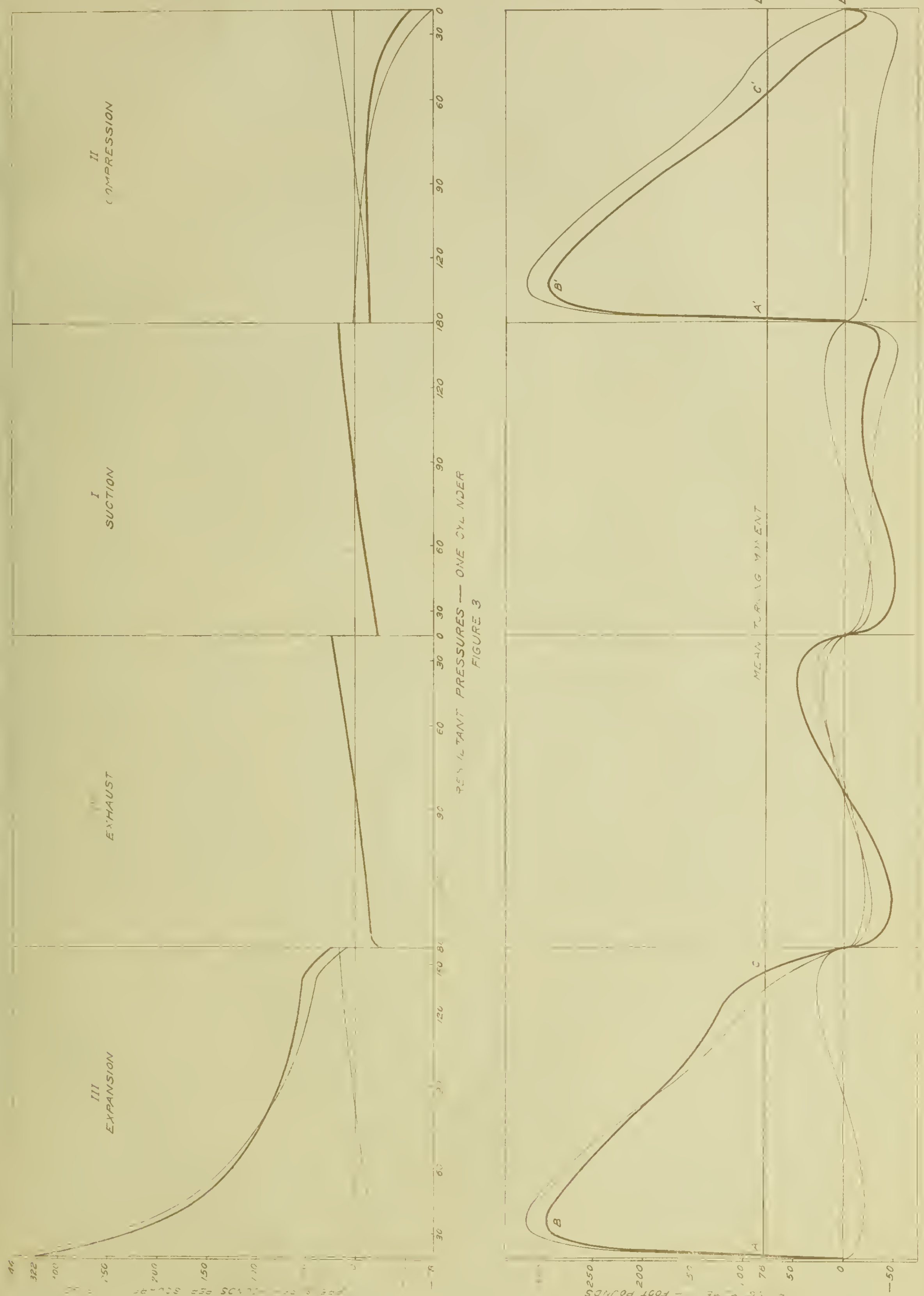
The derivation of this may be found in tests on gas engine design and will not be derived here. The tangential force is

$$P_R\left(\sin\theta + r/l\frac{\sin\theta\cos\theta}{1-(r/l)^2\sin^2}\right).$$

The turning moment is

$$P_R\left(\sin\theta + r/l\frac{\sin\theta\cos\theta}{1-(r/l)^2\sin^2\theta}\right)\times 2.5/12$$

PLATE II.


III
EXPANSION
EXHAUST
I
SUCTION
II
COMPRESSION
PRESSURES — ONE CYLINDER
FIGURE 3
MEAN TURNING MOMENT
FOOT POUNDS
A
B
C
D
E
A'
B'
C'

θ = crank angle.

P = total net force in pounds.

r/l = ratio of crank length to connecting rod length.

TABLE II.

θ	P lb.	Tang. Factor	Tang. Force lb.	Turning M. Ft.-lb.	θ	P lb.	Tang. Factor	Tang. Force Pounds	Turning M. Ft.-lb.
0	3800	0.	0	0	0	0	0.	0.	0.
15	3250	0.3157	1025	214	15	-232	0.3157	- 73.2	-15.3
30	2485	0.5992	1490	310	30	-205	0.5992	-122.8	-25.6
45	1825	0.8220	1500	312.5	45	-155	0.8220	-127.3	-26.5
60	1370	0.9664	1324	276	60	- 94	0.9664	- 90.8	-18.9
75	1060	1.0447	1108	231	75	- 28	1.0447	- 29.3	- 6.1
90	840	1.0000	840	175	90	33	1.0000	33	6.9
105	706	0.8871	626	130	105	83	0.8871	73.5	15.3
120	629	0.7656	481	100.2	120	127	0.7656	97.2	20.2
135	590	0.5922	349	72.7	135	160	0.5922	94.8	19.7
150	486	0.4008	195	40.5	150	171	0.4008	68.5	14.5
165	332	0.2019	67	14	165	177	0.2019	35.7	7.5
180	0	0.	0	0	180	0	0.	0.	0.
165	-210	0.2019	42.4	- 8.8	165	-166	0.2019	- 33.5	- 7.
150	-177	0.4008	71	-14.8	150	-177	0.4008	- 71	-14.8
135	-155	0.5922	91.8	-19.1	135	-155	0.5922	- 91.8	-19.1
120	-127	0.7656	-97.	-20.2	120	-155	0.7656	-118.7	-24.7
105	- 83	0.8871	-73.5	-15.3	105	-138	0.8871	-122.2	-25.5
90	- 33	1.0000	-33	- 6.9	90	-127	1.0000	-127	-26.4
75	28	1.0447	29.3	6.1	75	-133	1.0447	-139	-28.9
60	94	0.9664	91	19	60	-177	0.9664	-171	-35.6
45	155	0.8220	127.	26.4	45	-265	0.8220	-218	-45.4
30	205	0.5992	123	25.6	30	-409	0.5992	-245	-51
15	232	0.3157	73	15.2	15	-553	0.3157	-174.5	-35.4
0	0	0.	0	0	0	-620	0.	0.	0.

ARTICLE XIX.- THE CRANKSHAFT.- The maximum load comes on the crankshaft at starting. When running, the maximum pressure on the piston is reduced by the inertia forces. In considering bending stresses in the crankshaft the point of support is assumed to be 1/2 inch inside of the edge of the bearing and not at the center of the journal. These points are shown in Fig. 6 at E and F.

The reactions due to the pressure on the piston when the crank is on the upper dead center are:

$R_L = \dfrac{3800 \times 7.5}{10.25} = 2780$

pounds and

$R_R = \dfrac{3800 \times 2.75}{10.25} = 1020$

pounds.

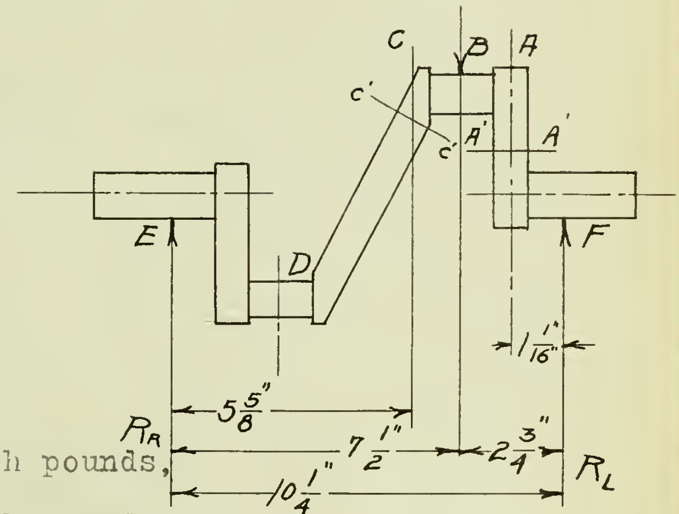


FIG. 6

The bending moments are:-

$M_A = 2780 \times 1.0625 = 2980$ inch pounds,

$M_B = 2780 \times 2.75 = 7650$ inch pounds,

$M_C = 1020 \times 5.625 = 5740$ inch pounds,

$M_D = 1020 \times 2.75 = 2800$ inch pounds.

The bending stresses are obtained as follows:-

$M_A = bd^2/6 \times S_A$, in which $bd^2/6 \times S_A$ = resisting moment at section A'A'.

$2980 = bd^2/6 \times S_A = \dfrac{2.125 \times 1}{6} \times S_A$

$S_A = 8420$ pounds per square inch.

$M_B = \pi d^3/32 \times S_B$; $d = 1.75$ inches.

$7650 = \dfrac{3.14 \quad 5.35}{32} \times S_B$

$S_B = 14,550$ pounds per square inch.

$M_c = bd^2/6 \times S_c = \frac{2.125 \times 1.563}{6} \times S_c$

$5740 = 0.554 \times S_c$

$S_c = 10{,}350$ pounds per square inch. The section resisting the bending moment, M_c, is section c'c' shown in Fig. 6 .

$M_D = \frac{\pi d^3}{32} \times S_D$; d = 1.75 inches.

$2800 = \frac{3.14 \times 5.35}{32} \times S_D.$

$S_D = 5340$ pounds per square inch.

The shearing stress at G, next to the crank arm, due to the load on the crank pin is $S_G = \frac{3800}{\pi d^2/4}$. With d = 1.75 inches the s shearing stress becomes 15,750 pounds per square inch.

The maximum turning moment as shown in Fig. 4 Plate II is 295 foot-pounds or 3540 inch pounds. This torque must be resisted by the crank shaft. The resisting moment of a solid shaft when subjected to twisting is $\pi d^3/16 \times S_s$. The $\pi d^3/16 \times S_s = 3540$ inch-pounds torque. With d = 1.75 inches, the shearing stress in the shaft is $S_S = \frac{3540 \times 16}{3.14 \times 5.35} = 3370$ pounds per square inch. This maximum stress occurs when the crank is about 35^0 past dead center.

The bending stress in the shaft due to the weight of the fly-wheel, about 100 pounds, is negligible. The effect of the over-hanging flywheel on the crank pins is to decrease the bending moments at these points. The actual bending stresses in the crank pins then are smaller than shown by the calculated stresses.

The allowable maximum pressure on the main journal is 400 to 550 pounds per square inch of projected area. The maximum load on the journal is the reaction or 2780 pounds. The diameter of

the journal is 1.75. Using a length of journal equal to 2d, the bearing area is 6.125 square inches and the maximum bearing pressure is 455 pounds per square inch. The weight of the flywheel, approximately 100 pounds, will have very little effect on the maximum bearing pressure.

ARTICLE XX.- THE FLYWHEEL.- A speed regulation of 4 percent will be close enough for the engine under consideration. The normal speed is 600 revolutions per minute or 10 revolutions per second. With 4 percent regulation the speed would vary from 9.8 to 10.2 revolutions per second, 2 percent above and 2 percent below normal speed.

It was determined from the drawing that the flywheel diameter could not exceed 15 inches. Assuming 2 inches as a depth of rim, the mean diameter of the rim is 13 inches.

The maximum velocity of the center of the whell rim is $V_{max.} = \frac{10.2\pi \times 13}{12} = 34.7$ feet per second, and the minimum velocity is $V_{min.} = \frac{9.8\pi \times 13}{12} = 33.35$ feet per second. The kinetic energy in foot-pounds given up by the flywheel by a reduction in speed is equal to $W/2g(V^2_{max.} - V^2_{min.})$ By integrating with a planimeter the turning moment curve shown on Plate II, page 19, it was found that the mean turning moment for a complete cycle of two revolutions was 78 foot-pounds and that the excess over the mean (area ABC plus Area A'B'C') was 208 foot-pounds. Then 104 foot-pounds energy must be absorbed and given up each revolution by the flywheel as its speed increases and decreases. Then

$104 = W/2g(V^2_{max.} - V^2_{min.})$, in which W is the weight of the flywheel rim and g the acceleration due to gravity. Substituting known values in the equation, $W = 104 \times 64.4/93 = 72$ pounds.

The area included between the outside and inside diameters of the rim is 81.7 square inches. The weight of the rim may be expressed as $81.7 \times f \times 0.26$, in which f is the face of the rim in inches and 0.26 the weight of cast iron per cubic inch. Then $f \times 81.7 \times 0.26 = 72$ and $f = 3.39$ inches. Making the face equal to 3 1/2 inches the weight of the rim becomes 74 pounds.

ARTICLE XXI.- THE GOVERNOR.- The engine will be throttled governed. The throttle lever on the carburetor will be operated thru links connected to a sliding collar on the crankshaft. This collar will be moved forward and backward on the shaft by bell-cranks connected to the fly-balls and pivoted on the fly-wheel. By adjusting the tension in the springs connecting the two fly-balls, the governor may be made to act at the desired rotative speed.

PLATE III